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2. Capillary tubes

- The capillary tube serves almost all small refrigeration systems, and its application extends up to refrigerating capacities of the order of 10 kW. A capillary tube is 1 to 6 m long with an inside diameter generally from 0.5 to 2 mm.
- The name is a misnomer, since the bore is too large to permit capillary action.
- Liquid refrigerant enters the capillary tube, and as it flows through the tube, the pressure drops because of friction and acceleration of the refrigerant. Some of the liquid flashes into vapor as the refrigerant flows through the tube.
- Numerous combinations of bore and length are available to obtain the desired restriction. Once the capillary tube has been selected and installed, however, the tube cannot adjust to variations in discharge pressure, suction pressure, or load.
- The compressor and expansion device must arrive at suction and discharge conditions which allow the compressor to pump from the evaporator the same flow rate of refrigerant that the expansion device feeds to the evaporator. A condition of unbalanced flow between these two components must necessarily be temporary.

- For a closer look at balance points the mass rate of flow fed by the capillary tube can be plotted on the same graph as the mass rate of flow pumped by the compressor. Figure 13-1 is such a plot with the flow through the capillary tube shown in dashed lines and the pumping capacity of a reciprocating compressor shown in solid lines. At high condensing pressures the capillary tube feeds more refrigerant to the evaporator than it does at low condensing pressures because of the increase in pressure difference across the tube.
- At high condensing pressures the capillary tube feeds more refrigerant to the evaporator than it does at low condensing pressures because of the increase in pressure difference across the tube.
- At a 30°C condensing temperature, for example, the compressor and capillary tube must search for a suction pressure which allows them both to pass equal mass rates of flow. This suction pressure is found at point 1, which is the balance point at a 30°C condensing temperature. Points 2 and 3 are the balance points at 40°C and 50°C condensing temperatures, respectively.

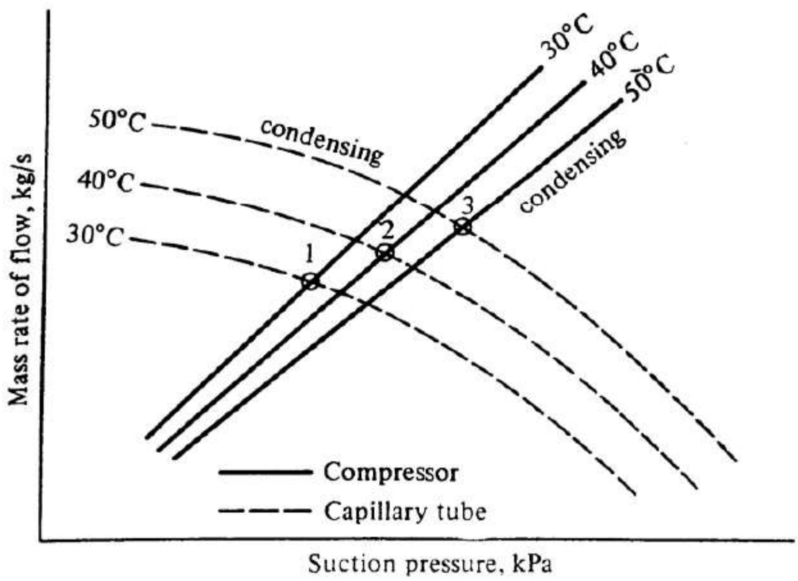

Figure 13-1 Balance points with a reciprocating compressor and capillary tube.

At suction pressure B the compressor can draw more refrigerant out of the evaporator than the capillary tube can supply, so the evaporator soon becomes short of refrigerant. Since the evaporator cannot be emptied indefinitely, something must happen to restore the balance. The corrective condition on most units without a receiver (vessel that stores liquid between the condenser and expansion device) is that liquid backs up into the condenser. The condensing area is thereby reduced and the condenser pressure raised. With the elevated condenser pressure, the compressor capacity is reduced and the capillary-tube rate of feed is increased until balance is restored. Another possibility for regaining a balanced flow rate is that the heat-transfer coefficient in the starved evaporator decreases. A greater temperature difference must develop between the fluid being chilled and the refrigerant in the evaporator, which occurs by means of the suction pressure dropping back to pressure A and restoring balanced flow.

An opposite unbalanced condition results if the refrigeration load falls off to less than the refrigeration capacity at the balance point. If the refrigeration load drops off, the suction temperature and pressure drop to some point C At suction pressure C the capillary tube can feed more refrigerant to the evaporator than the compressor can draw out. The evaporator fills with liquid and would spill over into the compressor with disastrous results were it not prevented. Slugging the compressor with liquid can be prevented by limiting the charge of refrigerant in the system. The charge is carefully measured so that there is enough refrigerant to fill the evaporator but no more. Balance of flow is restored when some gas enters the capillary tube, reducing the feed rate of the capillary tube 1 because of the high specific volume of the vapor. A new balance point is at point D in Fig. 13.2.

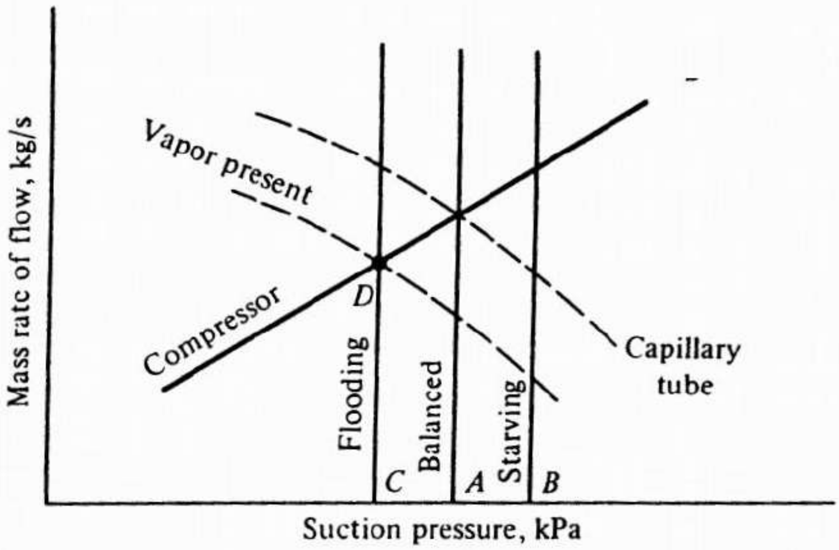

Figure 13-2 Unbalanced conditions causing starving or flooding of the evaporator. The condensing pressure is constant.

Although point *D* represents balanced flow, it is not a satisfactory condition. The state of the refrigerant entering the capillary tube shown on the pressure-enthalpy diagram in Fig. 13-3 is in the mixture region, which reduces the refrigerating effect compared with that when saturated or subcooled liquid enters the capillary tube. Each kilogram of refrigerant provides a reduced refrigerating effect in Fig. 13-3, but the work per kilogram remains unchanged.

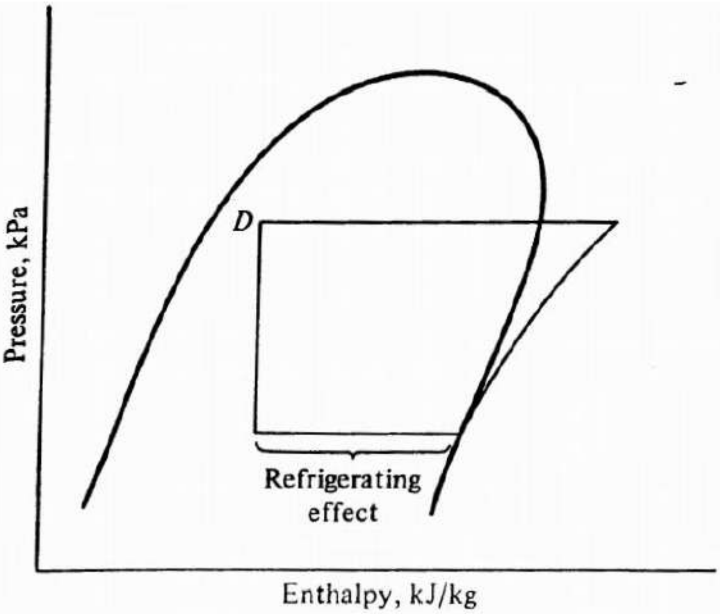

Figure 13-3 Reduction in refrigerating effect when some vapor enters the capillary tube.

3. Advantages and disadvantages of capillary tube

Capillary tubes have certain advantages and disadvantages.
Their advantages are summarized below;

1. Predominant enough to give them universal acceptance in factory-sealed systems.
2. They are simple, have no moving parts, and are inexpensive.
3. They also allow the pressures in the system to equalize during the off cycle.
4. The motor driving the compressor can then be one of low starting torque.

The disadvantages of capillary tubes are summarized below;

1. They are not adjustable to changing
2. Load conditions, are susceptible to clogging by foreign matter, and require the mass of
3. Refrigerant charge to be held within close limits

This last feature has dictated that the capillary tube be used only on hermetically sealed systems, where there is less likelihood of the refrigerant leaking out. The capillary tube is designed for one set of operating conditions, and any change in the applied heat load or condensing temperature from design conditions represents a decrease in operating efficiency.

Analytical computation of pressure drop in a capillary tube

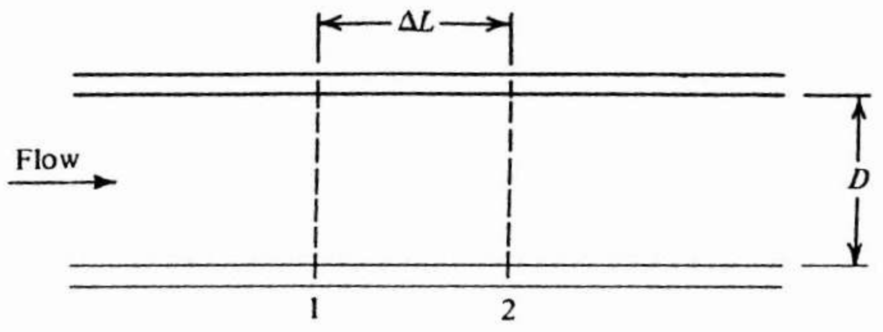

Figure 13-4 Incremental length of capillary tube.

The equations relating states and conditions at points I and 2 in a very short length of capillary tube in Fig. 13-4 will be written using the following notation:

A = cross-sectional area of inside of tube, m^2
D = ID of tube, m
f = friction factor, dimensionless
h = enthalpy, kJ/kg
h_f = enthalpy of saturated liquid, kJ/kg
h_g = enthalpy of saturated vapor, kJ/kg
ΔL = length of increment, m
p = pressure, Pa
Re = Reynolds number = $VD/\nu\mu$
ν = specific volume, m^3/kg
ν_f = specific volume of saturated liquid, m^3/kg
ν_g = specific volume of saturated vapor, m^3/kg
V = velocity of refrigerant, m/s
w = mass rate of flow, kg/s
x = fraction of vapor in mixture of liquid and vapor
μ = viscosity, Pa · s
μ_f = viscosity of saturated liquid, Pa · s
μ_g = viscosity of saturated vapor, Pa · s

The fundamental equations applicable to the control volume bounded by points 1 and 2 in Fig. 13-4 are (1) conservation of mass, (2) conservation of energy, and (3) conservation of momentum.

The equation for conservation of mass states that

$$w = \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2} \tag{13-1}$$

or

$$\frac{w}{A} = \frac{V_1}{v_1} = \frac{V_2}{v_2} \tag{13-2}$$

and w/A will be constant throughout the length of the capillary tube.

The statement of conservation of energy is

$$1000h_1 + \frac{V_1^2}{2} = 1000h_2 + \frac{V_2^2}{2} \tag{13-3}$$

which assumes negligible neat transfer in and out of the tube.

The momentum equation in words states that the difference in forces applied to the element because of drag and pressure difference on opposite ends of the element equals that needed to accelerate the fluid

$$\left[(p_1 - p_2) - f\frac{\Delta L}{D}\frac{V^2}{2v}\right]A = w(V_2 - V_1) \tag{13-4}$$

As the refrigerant flows through the capillary tube, its pressure and saturation temperature progressively drop and the fraction of vapor x continuously increases. At any point

$$h = h_f(1 - x) + h_g x \tag{13-5}$$

and

$$v = v_f(1 - x) + v_g x \tag{13-6}$$

In Eq. (13-4) V, v, and f all change as the refrigerant flows from point 1 to point 2, but some simplification results from Eq. (13-2), which shows that V/v is constant so that

$$f\frac{\Delta L}{D}\frac{V^2}{2v} = f\frac{\Delta L}{D}\frac{V}{2}\frac{w}{A} \tag{13-7}$$

In the calculation to follow in Example 13-1 the V used in Eq. (13-7) will be the mean velocity

$$V_m = \frac{V_1 + V_2}{2} \tag{13-8}$$

Since expressing the friction factor f for the two-phase flow is complex, we shall use an approximation and later compare the calculation with experimental results as a check on the validity of this approximation as well as of any other approximation built into the method.

For Reynolds numbers in the lower range of the turbulent region an applicable equation for the friction factor f is

$$f = \frac{0.33}{\mathrm{Re}^{0.25}} = \frac{0.33}{(VD/\mu v)^{0.25}} \tag{13-9}$$

The viscosity of the two-phase refrigerant at a given position in the tube is a function of the vapor fraction x

$$\mu = \mu_f(1 - x) + \mu_g x \tag{13-10}$$

The mean friction factor f_m applicable to the increment of length 1-2 is

$$f_m = \frac{f_1 + f_2}{2} = \frac{0.33/\mathrm{Re}_1^{0.25} + 0.33/\mathrm{Re}_2^{0.25}}{2} \tag{13-11}$$

13-5 Calculating the length of an increment The essence of the analytical calculation method is to determine the length of the increment 1-2 in Fig. 13-4 for a given reduction in saturation temperature of the refrigerant. The flow rate and all the conditions at point 1 are known, and for an arbitrarily selected temperature at point 2 the remaining conditions at point 2 and the ΔL will be computed in the following specific steps:

1. Select t_2.
2. Compute p_2, h_{f2}, h_{g2}, v_{f2}, and v_{g2}, all of which are functions of t_2.
3. Combine the continuity equation (13-2) and the energy equation (13-3)

$$1000h_2 + \frac{v_2^2}{2}\left(\frac{w}{A}\right)^2 = 1000h_1 + \frac{V_1^2}{2} \tag{13-12}$$

Substitute Eqs. (13-5) and (13-6) into Eq. (13-12)

$$1000h_{f2} + 1000(h_{g2} - h_{f2})x + \frac{[v_{f2} + (v_{g2} - v_{f2})x]^2}{2}\left(\frac{w}{A}\right)^2 = 1000h_1 + \frac{V_1^2}{2} \tag{13-13}$$

Everything in Eq. (13-13) is known except x, which can be solved by the quadratic equation

$$x = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \tag{13-14}$$

where

$$a = (v_{g2} - v_{f2})^2 \left(\frac{w}{A}\right)^2 \frac{1}{2}$$

$$b = 1000(h_{g2} - h_{f2}) + v_{f2}(v_{g2} - v_{f2}) \left(\frac{w}{A}\right)^2$$

$$c = 1000(h_{f2} - h_1) + \left(\frac{w}{A}\right)^2 \frac{1}{2} v_{f2}^2 - \frac{V_1^2}{2}$$

4. With the value of x known, h_2, v_2, and V_2 can be computed.
5. Compute the Reynolds number at point 2 using the viscosity from Eq. (13-10), the friction factor at point 2 from Eq. (13-9), and the mean friction factor for the increment from Eq. (13-11).
6. Finally, substitute Eqs. (13-7) and (13-8) into Eq. (13-4) to solve for ΔL.

Problem. 1: What length of capillary tube (ID = 1.63 mm) will drop the pressure of saturated liquid refrigerant 22 at 40°C to the saturation temperature of the evaporator of 5°C? The flow rate is 0.010 kg/s.

Use the following formula;

$$\ln\left(\frac{p}{1000}\right) = 15.06 - \frac{2418.4}{t + 273.15} \qquad (13\text{-}15)$$

$$v_f = \frac{v_f}{1000} = \frac{0.777 + 0.002062t + 0.00001608t^2}{1000} \qquad (13\text{-}16)$$

$$v_g = \frac{-4.26 + 94050(t + 273.15)/p}{1000} \qquad (13\text{-}17)$$

$$h_f = 200.0 + 1.172t + 0.001854t^2 \qquad (13\text{-}18)$$

$$h_g = 405.5 + 0.3636t - 0.002273t^2 \qquad (13\text{-}19)$$

$$\mu_f = 0.0002367 - 1.715 \times 10^{-6}t + 8.869 \times 10^{-9}t^2 \qquad (13\text{-}20)$$

$$\mu_g = 11.945 \times 10^{-6} + 50.06 \times 10^{-9}t + 0.2560 \times 10^{-9}t^2 \qquad (13\text{-}21)$$

Conditions at entrance to capillary tube, point 1 The entering refrigerant is saturated liquid at 40°C, and with $x = 0$ the properties from Eqs. (13-15) to (13-21) are

$p_1 = 1{,}536{,}000$ Pa $\quad v_1 = v_{f1} = 0.000885\ m^3/kg$
$h_1 = h_{f1} = 249.9$ kJ/kg $\quad \mu = \mu_{f1} = 0.0001823$ Pa · s

$$\frac{w}{A} = \frac{0.010}{\pi(0.00163^2)/4} = 4792.2\ kg/s \cdot m^2$$

$$V_1 = \frac{w}{A}v_1 = 4.242 m/s$$

$$Re_1 = 42{,}850 \qquad f_1 = \frac{0.33}{Re_1^{0.25}} = 0.0229$$

Conditions at point 2 Arbitrarily select $t_2 = 39$°C. Then

$p_2 = 1{,}498{,}800$ Pa $\quad h_{f2} = 248.5$ kJ/kg $\quad h_{g2} = 416.2$ kJ/kg
$v_{f2} = 0.000882\ m^3/kg$ $\quad v_{g2} = 0.01533\ m^3/kg$
$\mu_{f2} = 0.0001833$ Pa · s $\quad \mu_{g2} = 0.00001429$ Pa · s

From Eq. (13-14)

$x = 0.008$

From Eqs. (13-5) and (13-6) and using an equation of the same form for viscosity, we get

$h_2 = 249.84$ kJ/kg $\quad v_2 = 0.0009952$ m^3/kg
$\mu_2 = 0.0001820$ Pa ∘ s

The following terms can now be calculated:

$$V_2 = \frac{w}{A} v_2 = 4.769 \text{ m/s} \qquad \text{Re}_2 = 42{,}923$$

$$f_2 = \frac{0.33}{42{,}923^{0.25}} = 0.0229$$

$$f_m = \frac{0.0229 + 0.0229}{2} = 0.0229$$

$$V_m = \frac{4.242 + 4.769}{2} = 4.506$$

From Eq. (13-4) the magnitude of the expression

$$f_m \frac{\Delta L}{D} \frac{V_m}{2} \frac{V}{v}$$

is found to be 34,964, and when the known values are substituted,

$$\Delta L_{1-2} = 0.2306 \text{ m}$$

Table 13-1 Capillary-tube calculations in Example 13-1

Position	Temperature, °C	Pressure, kPa	x	Specific volume, m^3/kg	Enthalpy, kJ/kg	Velocity, m/s	Increment length, m	Cumulative length, m
1	40	1536.4	0.000	0.000885	249.85	4.242		
2	39	1498.8	0.008	0.000995	249.84	4.769	0.2306	0.231
3	38	1461.9	0.016	0.001110	249.84	5.320	0.2013	0.432
4	37	1425.8	0.023	0.001230	249.84	5.895	0.1770	0.609
5	36	1390.3	0.031	0.001355	249.83	6.496	0.1565	0.765
6–31	...	...	...	...	...	...	...	...
32	9	657.65	0.194	0.007660	249.18	36.71	0.0097	2.089
33	8	637.90	0.199	0.008048	249.11	38.57	0.0085	2.098
34	7	618.61	0.204	0.008452	249.03	40.51	0.0075	2.105
35	6	599.78	0.209	0.008873	248.95	42.52	0.0066	2.112
36	5	581.38	0.213	0.009309	248.86	44.61	0.0049	2.118

Details solution for the first two points had been inserted below;

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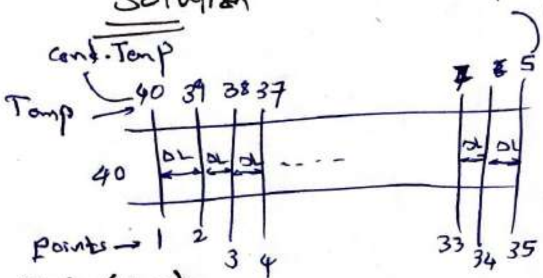
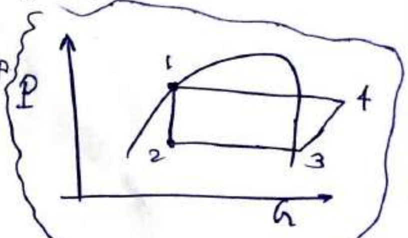
Example: What the length of Capillary tube (I.D = 1.63 mm) will drop the pressure of Saturated liquid refrigerant R-22 at 40 °C to the saturation temperature of the evaporator of 5 °C. Take the fluid flowrate as 0.010 Kg/sec.

Solution





Section (1-2):

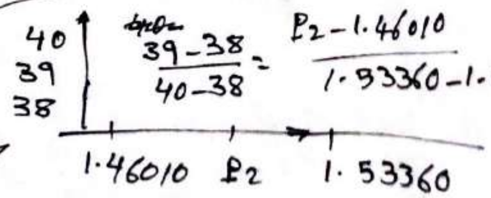
Momentum eq: $[(P_1 - P_2) - f \frac{\Delta L}{D} \frac{V^2}{2v}] A = \dot{m} (V_2 - V_1)$ ← should be Calculated

* The pressures can be easily Calculated From Refrigerant R-22 tables]

so: From table:

at $T_1 = 40\,°C \rightarrow P_1 = 1.53360$ MPa
$P_1 = 1.5336 * 10^6$ Pa
$T_2 = 39\,°C \rightarrow P_2 = 1.49685 * 10^6$ Pa

From Interpolation as illustrated here



$\frac{39-38}{40-38} = \frac{P_2 - 1.46010}{1.53360 - 1.46010} \rightarrow P_2$

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Friction factor; $f_m = \frac{f_1 + f_2}{2}$

Velocity; $V_m = \frac{V_1 + V_2}{2}$

Sp. Volume: $\forall_m = \frac{\forall_1 + \forall_2}{2}$

For point 1:

For Turbulent Flows: $f_1 = \frac{0.33}{Re_1^{0.25}}$

$$Re_1 = \frac{\rho_1 D V_1}{\mu_1} = \frac{D V_1}{\forall_1 \mu_1}$$

$D = 1.63$ mm

$V_1 \rightarrow \dot{m}_1 = \rho_1 V_1 A = \frac{V_1 A}{\forall_1}$

$$A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (1.63 * 10^{-3})^2$$

$$\therefore A = 2.08672438 * 10^{-6} m^2$$

$$\dot{m} = \frac{V_1 A}{\forall_1} ; \forall_1 = \forall_{f_1} (1 - x_1) + \forall_g x_1$$

$x_1 = 0$ (because state 1 lies at the sat. point)

$$\rightarrow \forall_1 = \forall_{f_1} = \frac{1}{\rho_{f_1}} = \frac{1}{1128.5} =$$

$$\therefore \forall_1 = 8.861320337 * 10^{-4} \ m^3/kg$$

In this way, V_1 can be calculated as:

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$$\dot{m} = \frac{V_1 A}{\nu_1} \rightarrow \dot{m}\,\nu_1 = V_1 A$$

$$V_1 = \frac{\dot{m}\,\nu_1}{A} = \frac{(0.010)(8.86132 * 10^{-4})}{2.086724 * 10^{-6}}$$

$$\therefore V_1 = 4.24652168\,8 \text{ m/s}$$

Two-Phase Dynamics Viscosity can be calculated:

$$\mu_1 = \mu_f (1 - x_1) + \mu_g x_1$$

$$\mu_1 = \mu_f = 139.4\ \mu Pa.s = 139.4 * 10^{-6}\ Pa.s$$

$$\therefore Re_1 = \frac{D V_1}{\nu_1 \mu_1} = \frac{1.63 * 10^{-3} * 4.2465}{8.86132 * 10^{-4} * 139.4 * 10^{-6}}$$

$$\therefore Re_1 = 56035.04913$$

$\because Re_1 > 2300$; then the fluid flow within the capillary tube will be turbulent

Thus;
$$f_1 = \frac{0.33}{Re^{0.25}} = \frac{0.33}{(56035.04913)^{0.25}}$$

$$\therefore f_1 = 0.0214486025$$

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For point 2

Since point 2 lies in the Mixture (Two-phase) region, it will be necessary to findout "x_2"

$$x_2 = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

$$a = (v_{g2} - v_{f2})^2 \left(\frac{\dot{m}}{A}\right)^2 \left(\frac{1}{2}\right)$$

From table at $T_2 = 39°C$

$$v_{f2} = 0.88048 * 10^{-3} \ m^3/kg$$

$$v_{g2} = 15.5375 * 10^{-3} \ m^3/kg$$

$$a = 2466.784353$$

$$b = 1000\,(h_{g2} - h_{f2}) + v_{f2}\,(v_{g2} - v_{f2}) \left(\frac{\dot{m}}{A}\right)^2$$

From Table: at $T_2 = 39°C$

$$h_{g2} = 416.388 \ kJ/kg.$$

$$h_{f2} = 248.361 \ kJ/kg.$$

$\therefore b = 168,323.3705$

$$c = 1000\,(h_{f2} - h_1) + \left(\frac{\dot{m}}{A}\right)^2 \frac{1}{2} v_{f2}^2 - \frac{V_1^2}{2}$$

$$c = -1289.114561$$

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In this way; $x_2 = 0.008$

$\therefore v_2 = v_{f_2}(1-x_2) + v_{g_2} x_2$

$= 0.88048 \times 10^{-3}(1-0.008) + \frac{15.5375}{10^3} \times 0.008$

$v_2 = 9.977 \times 10^{-4}\ m^3/kg$

$\mu_2 = \mu_{f_2}(1-x_2) + \mu_g x_2$

$= 0.0001820\ Pa.s$

$h_2 = h_{f_2}(1-x_2) + x_2 h_{fg_2}$

$h_2 = 249.84\ kJ/kg$

Since the Mass is Conserved,

$\dot{m}_2 = \dot{m}_1 = 0.01 = \rho_2 V_2 A = \frac{V_2 A}{v_2}$

$\therefore V_2 = \frac{\dot{m}\, v_2}{A} = 4.769\ m/s$

$Re_2 = \frac{\rho_2 D V_2}{\mu_2} = \frac{D V_2}{v_2 \mu_2}$

$Re_2 = 42923$

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$f_2 = \frac{0.33}{Re_2^{0.25}} = 0.0229$

$f_{m(1-2)} = \frac{0.021 + 0.0229}{2}$

$= 0.0229$

$V_m = \frac{4.242 + 4.769}{2} = 4.506$

From Eq. 13-4 by substitution the above calculated values:

$\Delta L_{1-2} = 0.2306\ m$

and use the same procedure with the others sections in order to find out ΔL_{2-3}, ΔL_{3-4} — — till we reach $T_{evap} = 5°C$ by summing all of ΔL

see the lecture to find the table of 13-1

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